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Comparative studies on performance evaluation of DI diesel engine with high grade low heat rejection combustion chamber with carbureted alcohols and crude jatropha oil

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A B S T R A C T

Search for renewable fuels such as vegetable oils and alcohols (ethanol and methanol) has become pertinent in the context of fossil fuel crisis and vehicle population explosion. The drawbacks associated with vegetable oils (high viscosity and low volatility) and alcohols (low cetane number) for use in diesel engines call for a hot combustion chamber, with its significant characteristics of higher operating temperature, maximum heat release, higher brake thermal efficiency and ability to handle the lower calorific value fuel. Investigations were carried out to evaluate the performance of a direct injection compression ignition engine with high grade low heat rejection (LHR) combustion chamber consisting of air gap insulated piston with 3 mm air gap with superni (an alloy of nickel) crown, air gap insulated liner with superni insert and ceramic coated cylinder head fueled with crude jatropha oil and carbureted alcohol (ethanol/methanol) with varied injection timings and injector opening pressures. Carbureted alcohol was inducted into the engine through a variable jet carburetor, installed at the inlet manifold of the engine at different percentages of crude vegetable oil at full load operation on mass basis. Comparative studies were made with engine with LHR combustion chamber with data of conventional engine with test fuels of diesel, crude vegetable oil and carbureted alcohol at recommended injection timing and optimized injection timing. Comparative studies were also made with methanol operation with data of ethanol operation on both versions of the combustion chamber with different operating conditions. Performance parameters, exhaust emissions and combustion characteristics were determined at full load operation of the engine with varied injection timings and injector opening pressures. Aldehydes were measured by the dinitrophenyl hydrazine (DNPH) method. Combustion diagnosis was carried out with a miniature piezoelectric pressure transducer, top dead center (TDC) encoder and special pressure–crank angle software package. The optimum injection timing was observed to be 32° bTDC with conventional engine while it was 29° bTDC for insulated engine with vegetable oil operation. The maximum induction of alcohol (methanol/ethanol) in conventional engine was found to be 35%, while it was 60% for the engine with LHR combustion chamber at recommended injection timing (27° bTDC). However, the maximum induction of alcohol was observed to be 55% with engine with LHR combustion chamber at its optimum injection timing. With maximum induction of methanol, at an injector opening pressure of 190 bar, engine with LHR combustion chamber at its optimum injection timing increased peak brake thermal efficiency by 3%; at full load operation brake specific energy consumption comparable, decreased exhaust gas temperature by 3%, decreased coolant load by 6%, volumetric efficiency comparable, increased formaldehyde levels by 30%, decreased acetaldehyde levels by 35%, decreased particulate emissions by 20%, decreased nitrogen oxide levels by 14%, increased peak pressures by 3% and maximum rate of pressure rose by 3%, when compared with ethanol operation on engine with LHR combustion chamber at its optimum injection timing.

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Contents

1. Introduction

When Rudolph Diesel, first invented the diesel engine, about a century ago, he demonstrated the principle by employing peanut oil and hinted that vegetable oil would be the future fuel in the diesel engine [1]. Several researchers experimented the use of vegetable oils as fuel on conventional engines (CE) and reported that the performance was poor, citing the problems of high viscosity, low volatility and their polyunsaturated character [2–8]. The high viscosity of the vegetable oils causes problems in the injection process leading to an increase in particulate emissions and the low volatility of the vegetable oils leads to oil sticking to the injector or combustion chamber walls resulting in deposit formation which interferes with the combustion. Increased injector opening pressures may also result in efficient combustion in compression ignition engine [6–8].

On the other hand, alcohols (ethanol and methanol) are renewable and volatile fuels. There are many methods of inducting alcohols in diesel engines, out of which blending is a simple technique [9–11]. However, the maximum amount of induction of alcohol in compression ignition engine is limited in the blending technique. Hence the carburetion technique of inducting alcohol in diesel engine is finding favor from various researchers from the point of view of effectiveness and operation. Methanol/ethanol was inducted through a variable jet carburetor, installed in inlet manifold of the engine and diesel was injected in conventional manner. Studies were done with carbureted methanol/ethanol with injected diesel fuel in conventional diesel engine [12–13]. It was reported from their investigations that performance improved with the carburetion technique. Exhaust emissions of particulate matter and nitrogen oxides (NO_x) decreased in comparison with pure diesel operation on conventional engine. However, ethanol/methanol has a low cetane number (less than 10). Hence engine modification is necessary if carbureted methanol/ethanol is used as fuel in diesel engine. The drawbacks associated with the crude vegetable oil and methanol/ethanol as fuels in diesel engine call for hot combustion chamber provided by LHR combustion chamber.

The second law requirement of Thermodynamics necessitates the inevitable heat loss to the coolant to realize work output. Any saving in this part of the energy distribution would either increase the energy lost through exhaust gases or increase the power output. Considerable efforts are under way to reduce heat loss to the coolant by various researchers. However, the results are a little confusing as to whether the insulation would improve or deteriorate thermal efficiency. The two approaches that are being pursued to decrease heat rejection are (1) ceramic coating and (2) air gap insulation. Engines with LHR combustion chamber are classified based on degree of insulation, such as low grade, medium grade and high grade insulated engines. Low grade LHR combustion chamber consists of ceramic coatings provided on engine components such as crown of the piston, inner side portion of the cylinder head and liner. Engine with medium grade LHR combustion chamber consists of two part piston – the top crown made of low thermal conductivity material is screwed to the aluminum body of the piston providing an air gap in between the crown and the body of the piston. An insert made of low thermal conductivity material is screwed to the top portion of the liner in such a manner that an air gap is maintained between the insert and liner body. Engine with high grade LHR combustion chamber is a combination of ceramic coated LHR combustion chamber provided with air gap insulation.

Investigations were carried out by various researchers on engine with low grade LHR ceramic coated combustion chamber with vegetable oil operation [14,15]. It was reported from their work that thermal efficiency increased by 2–3% and particulate emissions decreased by 20% with ceramic coated combustion chamber with vegetable oil operation in comparison with conventional combustion chamber with pure diesel operation.

Experiments were conducted on medium grade LHR combustion chamber with advanced injection timings and increased injector opening pressure with vegetable oils [16–19]. It was reported from their investigations that it improved thermal efficiency by 3–4%, decreased particulate matter by 25% and increased NO_x levels by 40% in comparison with pure diesel operation on conventional engine.

Nomenclature

ρ_a	density of air, kg/m^3
ρ_d	density of fuel, gm/cc
η_d	efficiency of dynamometer, 0.85
a	area of the orifice flow meter in square meter
BP	brake power of the engine, kW
C	number of carbon atoms in fuel composition
C_d	coefficient of discharge, 0.65
Cp	specific heat of water in kJ/kg K
D	bore of the cylinder, 80 mm
d	diameter of the orifice flow meter, 20 mm
DF	diesel fuel
H	number of hydrogen atoms in fuel
HSU	Hartridge smoke unit
I	ammeter reading, ampere
h	difference of water level in U-tube water manometer in cm of water column
IT	injection timing, degree bTDC
k	number of cylinders, 01
L	stroke of the engine, 110 mm
m_a	mass of air inducted in engine, kg/h
m_f	mass of fuel in kg/h
m_w	mass flow rate of coolant (water), kg/s
n	power cycles per minute, $N/2$,
N	speed of the engine, 1500 rpm
P_a	atmosphere pressure in mm of mercury
R	gas constant for air, 287 J/kg K
t	time taken for collecting 10 cc of fuel, s
T_a	room temperature, °C
T_I	inlet temperature of water, °C
T_o	outlet temperature of water, °C
V	voltmeter reading, V
V_s	stroke volume, m^3

Experiments were carried out with engine with high grade LHR combustion chamber with vegetable oils and reported performance deteriorated with conventional engine. Thermal efficiency increased by 4–5%, decreased particulate matter by 30% and increased NO_x emissions by 50% in comparison with pure diesel operation [20–22].

Carbureted alcohols (ethanol and methanol) were used in engine with high grade LHR combustion chamber and reported that performance improved with LHR combustion chamber and the effect of higher heat generated in the combustion space due to adiabatic conditions improved alcohol combustion with varying pilot quantities of vegetable oil [23–27].

Vegetable oils have high cetane number. Alcohol (ethanol and methanol) have high volatility and low C/H ratio. In order to obtain maximum performance from engine with minimum pollution levels, the combination of carbureted alcohol (ethanol/methanol) and crude vegetable oil as pilot fuel is to be attempted in compression ignition engine. No systematic studies on comparative performance of carbureted methanol and ethanol in engine with high grade LHR combustion chamber with varied injection timings and injector opening pressures were available.

An attempt was made here to evaluate the performance of the engine with high grade LHR combustion chamber, which contained air gap (3 mm) insulated piston with superni crown, air gap (3 mm) insulated liner with superni insert and ceramic coated cylinder head fueled with crude jatropha oil (CJO), with carbureted ethanol/methanol, with varied injector opening pressure and injection timing. Comparative studies were made with engine with LHR combustion chamber with data of conventional engine with test fuels of pure diesel, crude jatropha oil and carbureted ethanol/carbureted methanol with varied engine parameters.

2. Material and method

2.1. Fabrication of engine with LHR combustion chamber

Fig. 1 shows the assembly details of air gap insulated piston, air gap insulated liner and ceramic coated cylinder head. Engine with high grade LHR combustion chamber contained a two-part piston: the top portion, crown made of low thermal conductivity material (thermal conductivity of superni is 1/16 QUOTE of that of aluminum alloy of the piston) and superni 90, screwed to aluminum alloy body of the piston, with 3 mm air gap in between the crown and the body of the piston by providing gasket made of superni-90 material. An insert made of superni 90 was screwed to the top

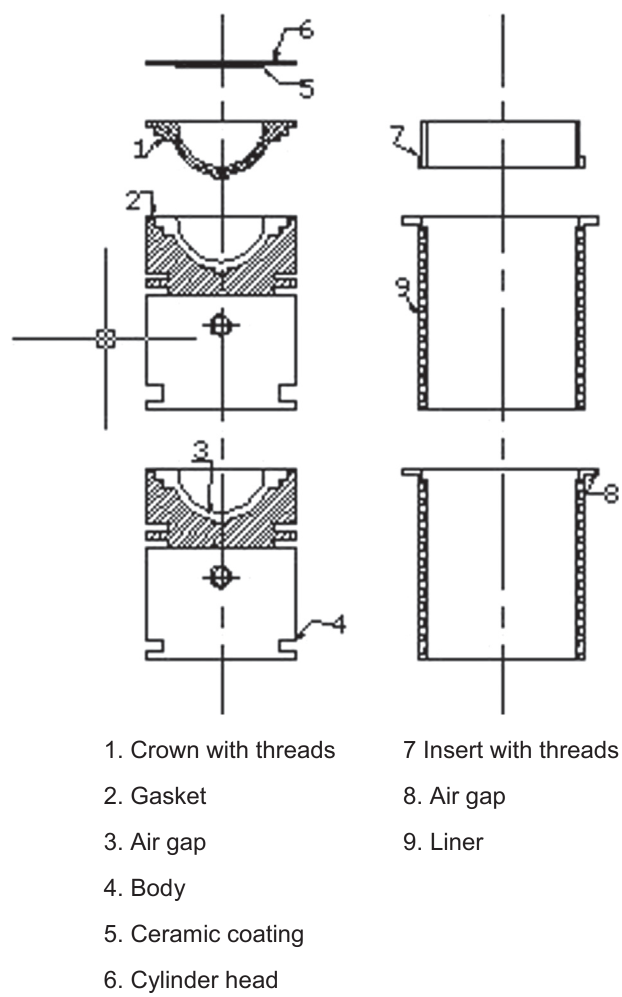

Fig. 1. Assembly details of air gap insulated piston, air gap insulated liner and ceramic coated cylinder head. (1) Crown with threads, (2) gasket, (3) air gap, (4) body, (5) ceramic coating, (6) cylinder head, (7) insert with threads, (8) air gap and (9) liner.

portion of the liner in such a manner that an air gap of 3 mm was maintained between the insert and the liner body [20]. Partially stabilized zirconium of thickness 500 microns was coated on the inside portion of cylinder head by the plasma spray technique.

2.2. Experimental setup

The schematic diagram of the experimental setup used for the investigations on the engine with LHR combustion chamber with jatropha oil and carbureted alcohol (ethanol/methanol) is shown in Fig. 2. The specifications of the experimental engine are shown in Table 1. The engine tests were carried out with a single cylinder, four stroke, naturally aspirated, compression ignition engine. The engine was operated at a constant speed of 1500 rev/min. The combustion chamber consisted of a direct injection type with no special arrangement for the swirling motion of air. The engine was connected to an electric dynamometer for measuring its brake power. Dynamometer was loaded by a loading rheostat. Brake power at different percentages of load was calculated by knowing the values of the output signals (voltmeter reading and ammeter reading) of dynamometer and speed of the engine. The accuracy of engine load is ± 0.2 kW. The speed of the engine was measured with digital tachometer with accuracy $\pm 1\%$.

The fuel consumption was registered with the aid of fuel measuring device (Burette and stop watch) and then mass flow rate of fuel was determined by knowing the density of the fuel. Density of fuel was determined by hydrometer. Percentage error obtained with measurement of fuel flow rate assuming laminar film in the burette was within the limit. The accuracy of determination of brake thermal efficiency obtained is $\pm 2\%$. A variable jet carburetor was fitted at the inlet manifold of the engine for inducting alcohol (methanol/ethanol) at different percentages at full load operation of vegetable oil on mass basis during the suction stroke of the engine. Crude jatropha oil was injected into the engine through conventional injection system. Two separate fuel tanks and glass burette arrangements were made for measuring vegetable oil and alcohol consumptions using stop watch. Bypass arrangement was provided for the engine to run with either pure vegetable oil/diesel fuel or carbureted alcohol along with vegetable oil.

Air-consumption of the engine was obtained with an aid of air box, orifice flow meter and U-tube water manometer assembly. By means of orifice flow meter and U-tube water manometer, discharge of air was calculated, from which mass flow rate of air was calculated. Percentage error obtained with measurement of difference of water levels in U-tube water manometer assuming laminar film in the manometer was within the limit. Air box with diaphragm was used to damp out the pulsations produced by the engine, for ensuring a steady flow of air through the intake manifold. The naturally aspirated engine was provided with water-cooling system in which inlet temperature of water was maintained at 80 °C by adjusting the water flow rate. The water flow rate was measured by means of analog water flow meter, with accuracy of measurement of $\pm 1\%$. Engine oil was provided with a pressure feed system. No temperature control was incorporated, for measuring the lube oil temperature. Copper shims of

Table 1
Specifications of the test engine.

Description	Specification
Engine make and model	Kirloskar (India) AV1
Maximum power output at a speed of 1500 rpm	3.68 kW
Number of cylinders × cylinder position × stroke	One × Vertical position × four-stroke
Bore × stroke	80 mm × 110 mm
Method of cooling	Water cooled
Rated speed (constant)	1500 rpm
Fuel injection system	In-line and direct injection
Compression ratio	16:1
BMEP @ 1500 rpm	5.31 bar
Manufacturer's recommended injection timing and pressure	27°bTDC × 190 bar
Dynamometer	Electrical dynamometer
Number of holes of injector and size	Three × 0.25 mm
Type of combustion chamber	Direct injection type
Fuel injection nozzle	Make: MICO-BOSCH No. 0431-202-120/HB
Fuel injection pump	Make: BOSCH NO. 8085587/1

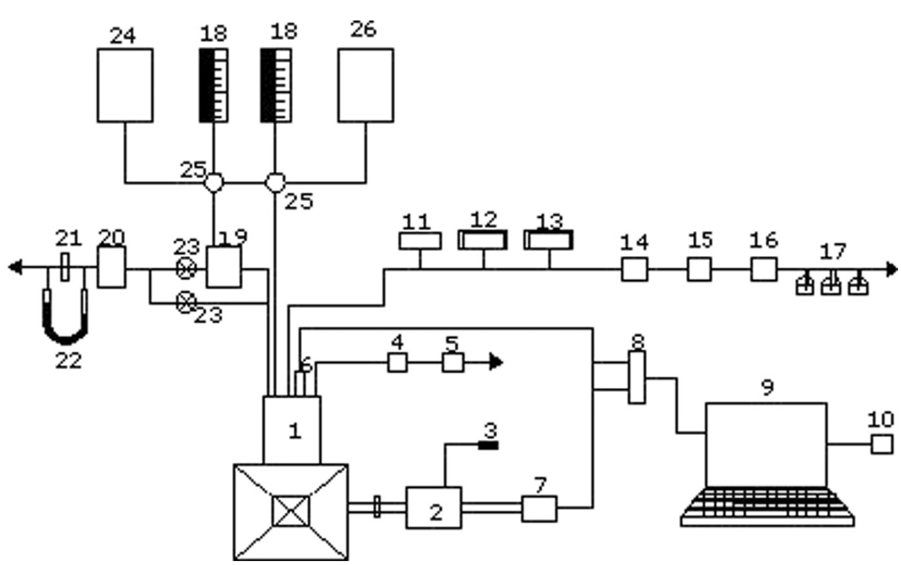

Fig. 2. Schematic diagram of the experimental set-up: (1) Engine, (2) electrical dynamometer, (3) load box, (4) outlet jacket water temperature indicator, (5) outlet-jacket water flow meter, (6) piezo-electric pressure transducer, (7) TDC encoder, (8) console, (9) Pentium personal computer, (10) printer, (11) exhaust gas temperature indicator, (12) AVL particulate matter analyzer, (13) Netel chromatograph NO_x analyzer, (14) filter, (15) rotometer, (16) heater, (17) round bottom flask containing DNPH solution, (18) burette, (19) variable jet carburetor, (20) air box, (21) orifice flow meter, (22) U-tube water manometer, (23) bypass valve, (24) alcohol tank, (25) three-way valve, and (26) vegetable oil tank.

suitable size were provided in between the pump body and the engine frame; to vary the injection timing and its effect on the performance of the engine it was studied, along with the change of injector opening pressure from 190 bar to 270 bar (in steps of 40 bar) using nozzle testing device. The maximum injector opening pressure was restricted to 270 bar due to practical difficulties involved. Exhaust gas temperature was measured by employing iron and iron–constantan thermocouples connected to a temperature indicator. The accuracy of analog exhaust gas indicator is ±1%.

2.3. Measurement of exhaust emissions

Exhaust emissions of particulate matter and NO_x were recorded by AVL (A company trade name) Particulate Matter Analyzer and Netel Chromatograph (A company trade name) analyzer at full load operation of the engine. The specifications of the analyzers are given in Table 2. The accuracy of analyzers is ±1%.

With alcohol–vegetable mixture operation, the major pollutant emitted from the engine is aldehydes. These aldehydes are carcinogenic in nature, which are harmful to human beings. The measure of the aldehydes is not sufficiently reported in the literature. The DNPH method was employed for measuring aldehydes in the experiment [23]. The exhaust of engine was filtered by means of filter and then heated up to 140 °C by means of heater provided in the circuit. A fixed quantity of exhaust (2l/m) measured by means of rotometer was bubbled through 2,4 dinitrophenyl hydrazine (2,4 DNPH) solution. The hydrazones formed were extracted into chloroform and were analyzed by employing high performance liquid chromatography (HPLC) test to find the percentage concentration of formaldehyde and acetaldehyde in the exhaust of the engine. The advantage of this method was determination of both formaldehyde concentration and acetaldehyde concentration simultaneously in the exhaust of the engine.

2.4. Determination of combustion characteristics

Piezo electric transducer, fitted on the cylinder head to measure pressure in the combustion chamber, was connected to a console, which in turn was connected to Pentium personal computer. TDC encoder provided at the extended shaft of the dynamometer was connected to the console to measure the crank angle of the engine. A special P–θ software package evaluated the combustion characteristics such as peak pressure, time of occurrence of peak pressure and maximum rate of pressure rise from the signals of pressure and crank angle at the full load operation of the engine. The accuracy of the instrumentation is ±1%.

2.5. Crude vegetable oil

Jatropha oil obtained from the plant *Jatropha curcas* plant has been found to be an attractive fuel in compression ignition engine. The *J. curcas* plant can be grown in arid and waste lands and needs very little attention. This oil is non-edible, and the plant is not even grazed by cattle. The whole seeds can be crushed to yield about 25% oil. Double crushing can increase the yield to 28.5% while solvent extraction to 30%.

Table 2
Specifications of analyzers.

Name of the analyzer	**Measuring range**	**Precision**	**Resolution**	**Accuracy of measurement**
AVL Particulate Matter Analyzer	0–100 HSU	1 HSU	1 HSU	±1%
Netel Chromatograph NO_x analyzer	0–2000 ppm	1 ppm	1 ppm	±1%

Table 3
Properties of test fuels.

Test fuel	Kinematic viscosity at 40 °C (mm^2/s)	Specific gravity 15 °C	Cetane number	Low calorific value (kJ/kg)
Diesel	3.07	0.84	55	42,000
Crude Jatropha oil	31.05	0.92	45	38,000
Ethanol	–	0.79	08	26,880
Methanol	–	0.81	03	19,740
ASTM	Standard	ASTM D 445	ASTM D 4809	ASTM D 613
ASTM D	4809			

2.6. Manufacturing of ethanol/methanol

Alcohol (methanol and ethanol) can be produced from organic materials such as grains, fruit, wood and even municipal solid wastes and waste or specifically grown biomass. The municipal solid wastes can be converted to alcohol. The wastes are first shredded and then passed under a magnet to remove ferrous materials. The iron free wastes are then gasified with oxygen. The product synthesis gas is cleaned by water scrubbing and other means to remove any particulates, entrained oils, H_2S and CO_2. CO-shift conversion for $H_2/CO/CO_2$ ratio adjustment, alcohol synthesis, and alcohol purification are accomplished. Alcohols (ethanol and methanol) are renewable in nature. They have oxygen in their molecular composition. They have low C/H value. They have low stoichiometric air–fuel ratio. Therefore, carbureted alcohol can be effectively used in compression ignition engine.

The properties of the test fuels of diesel, vegetable oil, ethanol and methanol used in this work are presented in Table 3. The low heating value of crude vegetable oil is approximately 10% lower than that of diesel fuel. However, specific gravity of crude vegetable oil is approximately 10% higher than that of diesel fuel. Hence energy content of vegetable oil is same as that of diesel fuel, with the same amount of fuel supplied.

2.7. Operating conditions

Test fuels used in the experimentation were diesel, crude vegetable oil along with carbureted ethanol/methanol. Different injector opening pressures attempted in this experiment were 190, 230 and 270 bar. Various injection timings attempted in the investigations were 27–34° bTDC. The various combustion chambers used in experiment were conventional combustion chamber and high grade LHR combustion chamber. The engine was started with diesel fuel and allowed to have a warm up for about 15 min. Each test was repeated ten times to ensure the reproducibility of data according to the procedure adopted in error analysis. (Minimum number of trials must be not less than ten.) The results were tabulated and comparative studies of performance parameters, exhaust emissions and combustion characteristics were reported at different operating conditions of the compression ignition engine.

2.8. Definitions of used values

$$m_f = \frac{10 \times \rho_d \times 3600}{t \times 1000} \tag{1}$$

$$BP = \frac{V \times I}{\eta_d \times 1000} \tag{2}$$

$$BTE = \frac{BP \times 3600}{m_f \times CV} \text{for diesel/vegetable oil operation} \tag{3}$$

$$BTE = \frac{BP \times 3600}{m_f \times CV + m_{al} \times CV_{al}} \text{ for alcohol \& vegetable oil operation}$$
$$m_{al} = \text{Mass of alcohol in kg/h}, CV_{al} = \text{Calorific value of alcohol} \tag{4}$$

$$BSEC = \frac{1}{BTE} \tag{5}$$

$$BP = \frac{BMEP \times 10^5 \times L \times A \times n \times k}{60000} \tag{6}$$

$$CL = m_w \times c_p \times (T_o - T_i) \tag{7}$$

$$m_a = C_d \times a \times \sqrt{2 \times 10 \times g \times h \times \rho_a} \times 3600 \tag{8}$$

$$\eta_v = \frac{m_a \times 2}{60 \times \rho_a \times N \times V_s} \tag{9}$$

$$\rho_a = \frac{P_a \times 10^5}{750 \times R \times T_a} \tag{10}$$

$$\% \text{ of alcohol inducted} = \frac{\text{mass of alcohol}}{\text{mass of alcohol} + \text{mass of vegetable oil at full load operation}} \tag{11}$$

Optimum injection timing: it is the injection timing at which maximum thermal efficiency of the engine is obtained at all loads.

2.9. Methodology adopted for carbureted alcohol with crude vegetable oil operation

Mass of vegetable oil at full load operation was determined at full load operation using Eq. (1). Mass of alcohol inducted into the engine was estimated knowing the percentage of alcohol induction and mass of the vegetable oil at full load operation using Eq. (11). Time taken for collecting 10 cc of alcohol was determined using Eq. (1). The knob of variable jet carburetor was adjusted so as to get time for collecting 10 cc of fuel into the engine by the trial and error method. Brake power at different loads of the engine was determined using Eq. (2). Brake thermal efficiency of the engine was calculated using Eq. (4). Brake specific energy consumption at full load operation was determined using Eq. (5). Brake mean effective pressure of the engine was determined using Eq. (6).

Coolant load was calculated using Eq. (7). Volumetric efficiency was determined using Eqs. (8)–(10).

3. Results and discussion

3.1. Performance parameters

3.1.1. Pure diesel operation

From Fig. 3, it is noticed that BTE increased up to 80% of the full load operation (4.2 bar) with pure diesel operation and beyond that load it decreased for both configurations of combustion chamber. The increase in efficiency of the engine was due to increase in fuel conversion efficiency. Decrease of efficiency beyond 80% of the full load was reduction of volumetric efficiency and oxygen–fuel ratio. Brake thermal efficiency increased with increasing injection timings at all loads, due to early initiation of combustion and increase of peak pressures. Maximum brake thermal efficiency was observed when the injection timing was advanced to 31° bTDC in conventional engine. Performance deteriorated for the injection timing greater than 31° bTDC. This was because of increase in ignition delay. Similar trends were observed in Reference [6].

From Fig. 4, it is noticed that brake thermal efficiency decreased at all loads in engine with LHR combustion chamber with pure diesel operation at the recommended injection timing when compared with CE. As the combustion chamber was insulated to greater extent, it was expected that high combustion temperatures

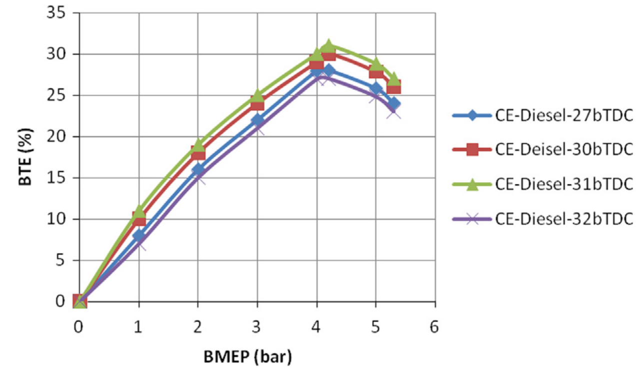

Fig. 3. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in conventional engine (CE) with pure diesel, at various injection timings at an injector opening pressure of 190 bar.

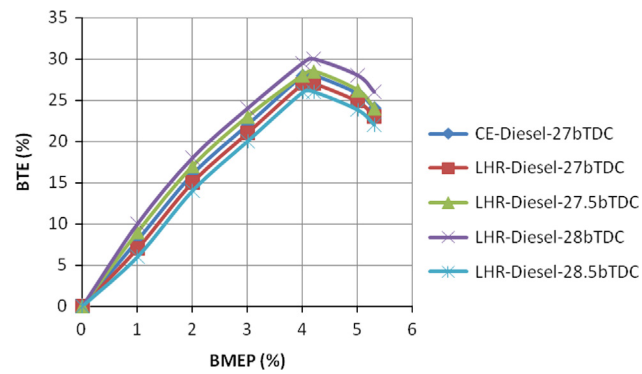

Fig. 4. Variation of brake thermal efficiency (BTE) in engine with LHR combustion chamber with pure diesel, at different injection timings and at an injector opening pressure of 190 bar.

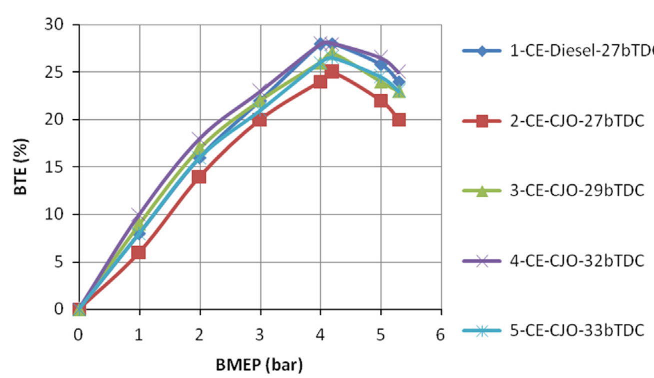

Fig. 5. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in conventional engine (CE) at different injection timings with crude jatropha oil (CJO) operation.

would be prevalent in engine with LHR combustion chamber. It tends to decrease the ignition delay thereby reducing pre-mixed combustion as a result of which, less time was available for proper mixing of air and fuel in the combustion chamber leading to incomplete combustion, with which brake thermal efficiency decreased at all loads. Moreover at this load, friction and increased diffusion combustion resulted from reduced ignition delay. Increased radiation losses might have also contributed to the deterioration. Similar observations were made in Reference [20]

Higher value of brake thermal efficiency at all loads including 100% full load was observed when the injection timing advanced to 28° bTDC in engine with LHR combustion chamber. Further advancing of the injection timing resulted in increase of fuel consumption due to longer ignition delay. Hence it was concluded that the optimized performance of engine with LHR combustion chamber was achieved at an injection timing of 28° bTDC with pure diesel operation.

Optimum injection timing (28° bTDC) was obtained earlier with engine with LHR combustion chamber in comparison with conventional engine (31° bTDC), as LHR combustion chamber of the engine was hotter.

3.1.2. *Crude vegetable oil operation*

From Fig. 5, it is observed that conventional engine with crude vegetable oil operation showed the deterioration in the performance at all loads when compared with the pure diesel operation on conventional engine at recommended injection timing. Crude jatropha oil has a low heating value because of the substantial amount of oxygen available in the fuel which is indicated by the fuel chemical formula given by $C_{18}H_{34}O_2$. In addition, crude vegetable oil is non-volatile in nature and it has high viscosity. These properties were responsible for the deterioration in the engine performance, which was confirmed by lower brake thermal efficiency. In addition, less oxygen entrainment by the fuel spray suggested that the fuel spray penetration might increase and result in more fuel reaching the combustion chamber walls. Furthermore droplet mean diameters (expressed as Sauter mean) were larger for vegetable oil leading to reduction in the rate of heat release as compared with diesel fuel [7].

Brake thermal efficiency increased with the advancement of the injection timing in CE with vegetable oil at all loads, when compared with conventional engine at recommended injection timing and injector opening pressure. This was due to initiation of combustion at earlier period and efficient combustion with increase of peak pressure with increase of oxygen entrainment in fuel spray giving higher brake thermal efficiency. Brake thermal efficiency increased at all loads when the injection timing was advanced to 32° bTDC in conventional engine at the normal temperature of vegetable oil. The increase of BTE at optimum injection timing over the recommended injection timing with vegetable oil with CE was attributed to its longer ignition delay and combustion duration. Similar trends were observed in Reference [22].

From Fig. 6, it is noticed that, engine with LHR combustion chamber with vegetable oil operation showed the improvement in the performance at all loads compared with conventional engine with pure diesel operation. High cylinder temperatures helped in improved evaporation and faster combustion of the fuel injected into the combustion chamber. Similar trends were noticed in Reference [22].

Reduction of ignition delay of the vegetable oil in the hot environment of the engine with LHR combustion chamber improved heat release rates and efficient energy utilization. The optimum injection timing was found to be 29° bTDC with engine with LHR combustion chamber with crude vegetable oil.

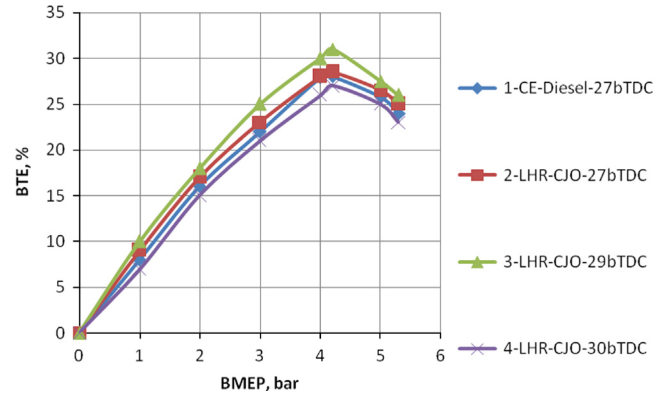

Fig. 6. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in engine with LHR combustion chamber at different injection timings with crude jatropha oil (CJO) operation.

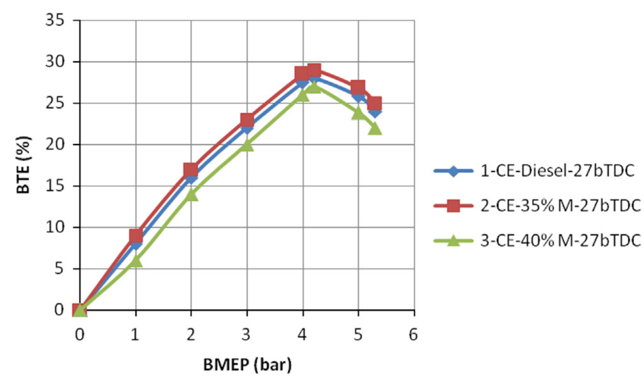

Fig. 7. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in conventional engine (CE) at different percentages of methanol (M) induction at 27° bTDC and injector opening pressure of 190 bar.

The hot combustion chamber of engine with LHR combustion chamber reduced ignition delay and combustion duration; hence the optimum injection timing was obtained earlier with engine with LHR combustion chamber when compared with conventional engine with the vegetable oil operation.

3.1.3. *Carbureted alcohol operation*

Investigations were carried out with the objective of determining the factors that would allow maximum induction of alcohol in conventional engine and engine with LHR combustion chamber with best possible efficiency at all loads.

Fig. 7 indicates that brake thermal efficiency increased at all loads with 35% methanol (M) induction and with an increase of methanol induction beyond 35%, it decreased at all loads in CE when compared with conventional engine with diesel operation. The reason for improving thermal efficiency with the 35% methanol induction was improved homogeneity of the mixture with the presence of methanol, decreased dissociated losses, specific heat losses and cooling losses due to lower combustion temperatures. This was also due to high heat of evaporation of methanol, which caused reduction in the gas temperatures resulting in a lower ratio of specific heats leading to more efficient conversion of heat into work. Induction of methanol resulted in more moles of working gas, which caused high pressures in the cylinder. The observed increase in the ignition delay period would allow more time for fuel to vaporize before ignition started. This means higher burning rates resulted in more heat release rate at

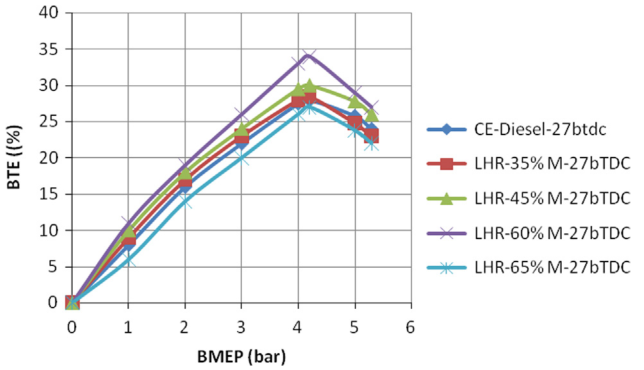

Fig. 8. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in engine with LHR combustion chamber at different percentages of methanol (M) induction at 27° bTDC and injector opening pressure of 190 bar.

constant volume, which was a more efficient conversion process of heat into work. Similar observations were made in References [23–27].

Curves in Fig. 8 indicate that engine with LHR combustion chamber showed an improvement in the performance with the carbureted methanol at all loads when compared with the conventional engine with diesel operation. This was due to recovery of heat from the hot insulated components of engine with LHR combustion chamber due to high latent heat of evaporation of methanol. The maximum induction of methanol was 60% in engine with LHR combustion chamber, which showed improvement in the performance at all loads when compared with diesel operation on CE.

However when the methanol induction was increased more than 60% in engine with LHR combustion chamber, brake thermal efficiency deteriorated at all loads when compared with conventional engine with diesel operation. This was due to increase of ignition delay. Similar observations were noticed with carbureted ethanol with crude vegetable oil operation.

The optimum injection timing was at 31° bTDC for conventional engine, while it was 28° bTDC for LHR engine with pure diesel operation. In case of crude vegetable oil operation, optimum injection timing was at 32° bTDC for conventional engine, while it was 29° bTDC for engine with LHR combustion chamber. However, maximum induction of alcohol was limited to 55% in the engine with LHR combustion chamber at 29° bTDC against 60% induction at 27° bTDC, while maximum induction of alcohol remained the same in conventional engine at 32° bTDC as in the case of 27° bTDC.

3.2. Comparative studies with test fuels with conventional engine and engine with LHR combustion chamber

3.2.1. Performance parameters

The part load variations of the parameters with respect to brake mean effective pressure were small; hence bar charts were drawn at full load operation on conventional engine and engine with LHR combustion chamber with pure diesel, crude vegetable oil and carbureted methanol at recommended and optimum injection timings at recommended injector opening pressure of 190 bar. With the help of tables, comparative studies were made with engine with LHR combustion chamber with data of conventional engine with the test fuels at different operating conditions.

From Fig. 9, it is noticed that engine with LHR combustion chamber with 55% methanol induction at its optimum injection timing showed improved peak brake thermal efficiency when compared with other versions of the combustion chamber. This was due to higher amount of methanol substitution and improved combustion at advanced injection timing which caused improved evaporation leading to produce higher brake thermal efficiency.

Conventional engine with crude jatropha oil operation gave lower brake thermal efficiency when compared with other versions of the combustion chamber. This was because of high viscosity, low volatility and low calorific value of the crude vegetable oil. Similar observations were made in Reference [22].

There is a limitation to use methanol in CE due to its low cetane number and high self-ignition temperature than vegetable oil without increasing injector opening pressure. As percentage of methanol increased, more heat was utilized to evaporate alcohol fuel and less heat was available to evaporate vegetable oil. Therefore a major quantity of alcohol which burned late in the expansion stroke, would not be fully utilized. In order to avert this, injector opening pressure was increased, which reduced droplet

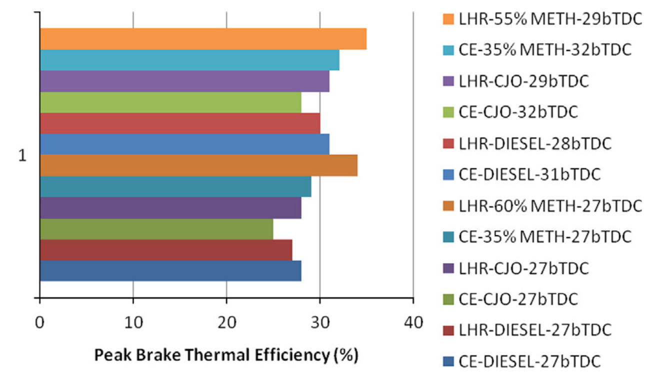

Fig. 9. Bar charts showing the variation of peak brake thermal efficiency (BTE) with test fuels in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

Table 4
Comparative data on peak brake thermal efficiency.

IT	Combustion chamber version	Alcohol induction on mass basis	Peak brake thermal efficiency (%)					
			Methanol operation			Ethanol operation		
			Injector opening pressure (bar)			Injector opening pressure (bar)		
			190	230	270	190	230	270
27	CE	0% (DF)	28	29	30	28	29	30
		0% (CJO)	25	26	27	25	26	27
		35%	29	30	31	29.5	30	30.5
		40%	–	–	32	–	–	32.2
	LHR	0% (DF)	27	28	29	27	28	29
		0% (CJO)	28	29	30	28	29	30
		60%	34	35.5	36	33	33.5	34
28	LHR	0% (DF)	30	31	31.5	30	31	31.5
29	LHR	0% (CJO)	31	31.5	32	31	31.5	32
	LHR	55%	35	36	36.5	34	34.5	35
31	CE	0% (DF)	31	30.5	30	31	30.5	30
32	CE	0% (CJO)	28	27	26	28	27	26
32	CE	35%	32	32.5	33	32.2	32.5	33

size of fuel, increased surface to volume ratio and required comparatively less heat to evaporate vegetable oil droplet.

The trend exhibited by both versions of the combustion chamber with dual fuel operation at higher injector opening pressure of 270 bar was similar to the corresponding injector opening pressure of 190 bar as noticed from Table 4. However, the maximum induction of alcohol was 40% in conventional engine at an injector opening pressure of 270 bar against 35% at 190 bar, while maximum alcohol induction remained same with engine with LHR combustion chamber at 270 bar as in the case of 190 bar. This was due to improved gas temperatures with CE and decrease of the same with engine with LHR combustion chamber.

From Table 4, it is noticed that with pure diesel operation, engine with LHR combustion chamber decreased peak brake thermal efficiency by 4% at recommended injection timing and 3% at optimized injection timing in comparison with conventional engine. This was due to reduction of ignition delay. Similar observations were made in Reference [22].

With vegetable oil operation, peak brake thermal efficiency increased by 12% at recommended injection timing and 11% at optimized injection timing with engine with LHR combustion chamber in comparison with conventional engine. This was due to improved combustion in the hot environment provided by LHR combustion chamber. Similar observations were made in References [22,23]. However, peak brake thermal efficiency of the engine with LHR combustion chamber obtained by reference was higher than that of engine with LHR combustion chamber employed by authors [23]. This was due to high calorific value of the fuel (pongamia oil) employed in Reference [23].

Engine with LHR combustion chamber with 60% methanol induction increased peak BTE by 17% at recommended injection timing and 9% at optimized injection timing in comparison with conventional engine with 35% methanol induction. This was due to higher amount of methanol substitution and improved evaporation rate in the engine with LHR combustion chamber. Similar trends were observed in Reference [24].

Peak brake thermal efficiency was marginally higher with conventional engine with ethanol operation in comparison with methanol operation on conventional engine at recommend injection timing and optimized injection timing. This was due to higher calorific value of ethanol when compared with methanol.

However, methanol operation on engine with LHR combustion chamber improved the performance when compared with ethanol operation on the same version of the engine. This was because of higher evaporation characteristics of methanol in the hot environment provided by engine with LHR combustion chamber.

Peak brake thermal efficiency increased with an increase of injector opening pressure in both versions of the combustion chamber with test fuels. This was due to improved spray characteristics of the fuel with increased injector opening pressure.

As fuels with different calorific values were used in the investigations, brake specific energy consumption (BSEC), defined as energy supplied through the fuel per unit power output of the engine, was used instead of brake specific fuel consumption (BSFC), defined as fuel consumed per unit brake power.

From Fig. 10, it is noticed that engine with LHR combustion chamber with 55% methanol induction at its optimum injection timing showed lower BSEC at full load when compared with other versions of the combustion chamber with other test fuels. This was due to higher amount of methanol substitution and improved combustion at advanced injection timing that caused improved evaporation leading to produce lower brake specific energy consumption.

Conventional engine with vegetable oil operation increased BSEC at full load operation, when compared with other versions of the combustion chamber with other test fuels. This was because of high viscosity, poor volatility and reduction in heating value of vegetable oil that lead to their poor atomization and combustion characteristics. Similar trends were noticed in Reference [6]

BSEC at full load operation decreased with advanced injection timing with test fuels. This was due to initiation of combustion at early period.

From Table 5, it is noticed that, with pure diesel operation, engine with LHR combustion chamber increased BSEC at full load operation by 5% at recommended injection timing and 2% at optimized injection timing in comparison with conventional engine. This was due to reduction of ignition delay with less time available for complete combustion. Similar trends were noticed in Reference [20].

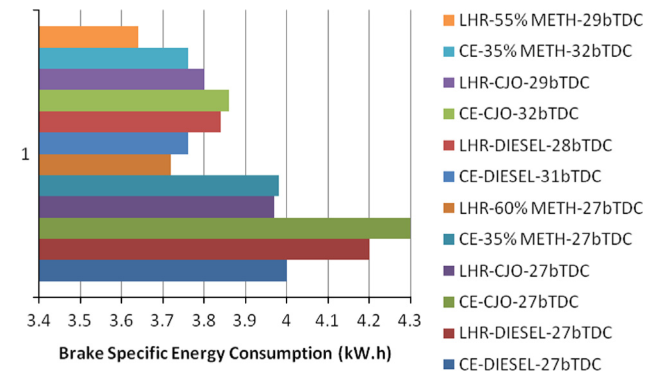

Fig. 10. Bar charts showing the variation of brake specific energy consumption (BSEC) at full load operation with test fuels in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

Table 5
Comparative data on brake specific energy consumption at full load operation

IT	Combustion chamber version	Alcohol induction on mass basis	Brake specific energy consumption (kW h)					
			Methanol operation			Ethanol operation		
			Injector opening pressure (bar)			Injector opening pressure (bar)		
			190	230	270	190	230	270
27	CE	0% (DF)	4.0	3.92	3.84	4.0	3.92	3.84
		0% (CJO)	4.30	4.26	4.22	3.97	3.93	3.89
		35%	3.98	3.96	3.94	3.88	3.86	3.84
		40%	–	—	3.88	–	—	3.76
	LHR	0% (DF)	4.2	4.16	4.12	4.2	4.16	4.12
		0% (CJO)	3.97	3.93	3.89	3.97	3.93	3.89
		60%	3.72	3.70	3.68	3.74	3.72	3.70
28	LHR	0% (DF)	3.84	3.80	3.76	3.84	3.80	3.76
29	LHR	0% (CJO)	3.80	3.76	3.72	3.80	3.76	3.72
	LHR	55%	3.64	3.62	3.60	3.68	3.66	3.64
31	CE	0% (DF)	3.76	3.80	3.84	3.76	3.80	3.84
32	CE	0% (CJO)	3.86	3.90	3.94	3.86	3.90	3.94
32	CE	35%	3.76	3.74	3.72	3.75	3.73	3.71

From the same table, it is evident that, with vegetable oil operation, BSEC at full load operation decreased by 8% at recommended injection timing and 2% at optimized injection timing with engine with LHR combustion chamber in comparison with conventional engine. This was due to high viscosity, low volatility and low calorific value of the vegetable oil leading to deterioration in performance in conventional engine. Similar observations were made in Reference [22].

With maximum induction of methanol, BSEC at full load operation decreased by 7% at recommended injection timing and 3% at optimized injection timing with LHR combustion chamber in comparison with conventional engine. This was due to high amount of methanol substitution in the engine with LHR combustion chamber. Similar trends were noticed in Reference [24].

BSEC at full load operation decreased with the increase of alcohol induction, as higher amount of alcohol substitution caused improved evaporation and produced lower BSEC in both versions of the combustion chamber.

BSEC was lower in engine with LHR combustion chamber with carbureted alcohol at its optimum injection timing, which shows the suitability of the engine for alternative fuels.

BSEC was lower with conventional engine with ethanol induction, while it was lower with engine with LHR combustion chamber with methanol induction at recommended injection timing and optimized injection timing. This was due to high calorific value of ethanol and high latent heat of evaporation of methanol.

BSEC at full load operation decreased with an increase of injector opening pressure and advanced injection timing in both versions of the combustion chamber with test fuels. This was due to decrease of mean diameter of the droplet with increased injector opening pressure.

Fig. 11 indicates that, engine with LHR combustion chamber with 55% methanol induction at its optimum injection timing showed lower value of exhaust gas temperature at full load operation when compared with other versions of the combustion chamber with other test fuels. This was due to high latent heat of evaporation of methanol which absorbed gas temperatures leading to reduction in exhaust gas temperature.

Conventional engine with vegetable oil operation gave higher value of exhaust gas temperature when compared with other versions of the combustion chamber with test fuels. Though calorific value (or heat of combustion) of vegetable oil was less than that of diesel, density of the vegetable oil was higher, therefore greater amount of heat was released in the combustion chamber leading to higher exhaust gas temperature with conventional engine, which confirmed that performance deteriorated with conventional engine with vegetable oil operation in comparison with pure diesel operation. This was also because of high duration of combustion of vegetable oil causing retarded heat release rate. Similar findings were obtained by other studies [22].

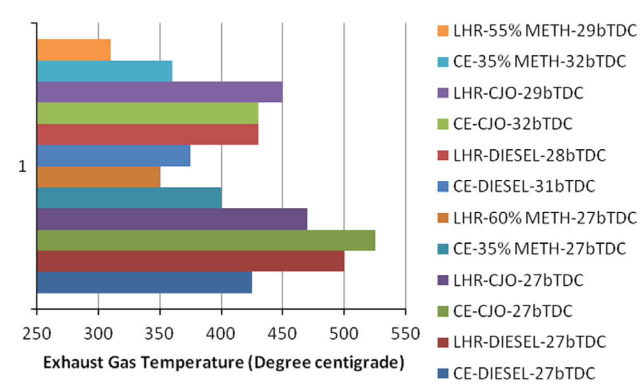

Fig. 11. Bar charts showing the variation of exhaust gas temperature (EGT) with test fuels in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

Table 6
Comparative data on exhaust gas temperature (EGT) at full load operation.

IT	Combustion chamber version	Alcohol induction on mass basis	Exhaust gas temperature (°C)					
			Methanol operation			Ethanol operation		
			Injector opening pressure (bar)			Injector opening pressure (bar)		
			190	230	270	190	230	270
27	CE	0% (DF)	425	410	395	425	410	395
		0% (CJO)	525	500	475	525	500	475
		35%	400	375	350	375	350	325
		40%	–	–	320	–	–	300
	LHR	0% (DF)	500	475	450	500	475	450
		0% (CJO)	470	440	410	470	440	410
		60%	350	325	300	360	340	320
28	LHR	0% (DF)	430	410	390	430	410	390
29	LHR	0% (CJO)	450	420	390	450	420	390
	LHR	55%	310	290	270	320	300	280
31	CE	0% (DF)	375	400	425	375	400	425
32	CE	0% (CJO)	430	450	470	430	450	470
32	CE	35%	360	340	320	340	320	300

Exhaust gas temperatures decreased with advanced injection timing with test fuels as seen from Fig. 11. This was because, when the injection timing was advanced, the work transfer from the piston to the gases in the cylinder at the end of the compression stroke was too large, leading to reduction in exhaust gas temperatures.

From Table 6, it is noticed that, with pure diesel operation, the exhaust gas temperature at full load operation increased by 18% at recommended injection timing and 15% at optimized injection timing with engine with LHR combustion chamber, in comparison with conventional engine. This indicated that heat rejection was restricted through the piston, liner and cylinder head, thus maintaining the hot combustion chamber as result of which exhaust gas temperature increased with reduction of ignition delay. Similar observations were made in Reference [22].

From the same table, it is observed with vegetable oil operation, exhaust gas temperature at full load operation decreased by 10% at recommended injection timing and increased by 5% at optimized injection timing with engine with LHR combustion chamber in comparison with conventional engine. This showed that combustion improved with engine with LHR combustion chamber at recommended injection timing. Exhaust gas temperature decreased with conventional engine in comparison with engine with LHR combustion chamber due to its higher advanced injection timing. Similar trends were noticed in Reference [22].

From the table, it is further evident that, with maximum induction of methanol, exhaust gas temperature at full load operation decreased by 13% at recommended injection timing and 14% at optimized injection timing, with engine with LHR combustion chamber in comparison with conventional engine. This was because of higher amount of methanol induction in

engine with LHR combustion chamber leading to reduction in exhaust gas temperature, due to its high latent heat of evaporation. Similar trends were noticed in Reference [24].

Exhaust gas temperature decreased with the increase of percentage of alcohol induction in both versions of the combustion chamber. Exhaust gas temperature reduced in both versions of the combustion chamber with methanol induction in comparison with ethanol induction at the recommended and optimized injection timings. This was because of high latent heat of evaporation of methanol in comparison with ethanol.

Exhaust gas temperature decreased marginally with an increase of injector opening pressure in both versions of the combustion chamber. This is due to improved spraying characteristics of the fuel with an increase of injector opening pressure.

From Fig. 12, it is observed that that engine with LHR combustion chamber with 55% methanol induction at its optimum injection timing showed lower value of coolant load at full load operation when compared with other versions of the combustion chamber with other test fuels. This was due to high latent heat of evaporation of methanol which absorbed gas temperatures leading to reduction in coolant load along with provision of thermal insulation.

Conventional engine with vegetable oil operation gave higher value of coolant load when compared with other versions of the combustion chamber with other test fuels. This was due to increase of un-burnt fuel concentration at the combustion chamber walls with deterioration in combustion with conventional engine leading to produce high value of coolant load.

In case of conventional engine, un-burnt fuel concentration reduced with effective utilization of energy, released from the combustion, coolant load with test fuels increased marginally at full load operation, with increase of gas temperatures, when the injection timing was advanced to the optimum value. Coolant load decreased with advanced injection timing with engine with LHR combustion chamber with test fuels. This was due to decrease of gas temperatures with effective combustion. However, the improvement in the performance of the conventional engine was due to heat addition at higher temperatures and rejection at lower temperatures, while the improvement in thermal efficiency of the engine with LHR combustion chamber was due to heat recovery from coolant load at optimum injection timings with test fuels.

From Table 7, it is noticed that with pure diesel operation, coolant load at full load operation decreased by 5% at recommended injection timing and 14% at optimized injection timing with engine with LHR combustion chamber, in comparison with conventional engine. This was due to the provision of thermal insulation with engine with LHR combustion chamber. This was also due to reduction of gas temperatures with LHR combustion

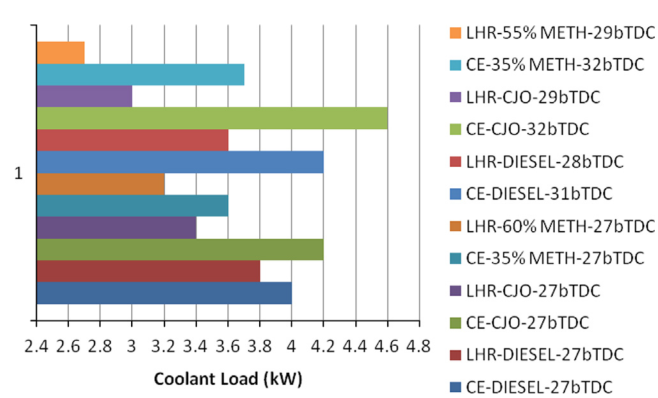

Fig. 12. Bar charts showing the variation of coolant load with test fuels in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

Table 7
Comparative data on coolant load at full load operation.

IT	Combustion chamber version	Alcohol induction on mass basis	Coolant load (kW)					
			Methanol operation			Ethanol operation		
			Injector opening pressure (bar)			Injector opening pressure (bar)		
			190	230	270	190	230	270
27	CE	0% (DF)	4.0	4.2	4.4	4.0	3.8	3.6
		0% (CJO)	4.2	4.4	4.6	4.2	4.4	4.6
		35%	3.6	3.8	4.0	3.7	3.9	4.1
		40%	–	–	3.8	–	–	3.9
	LHR	0% (DF)	3.8	3.6	3.4	3.8	3.6	3.4
		0% (CJO)	3.4	3.2	3.0	3.4	3.2	3.0
		60%	3.2	3.0	2.8	3.4	3.2	3.0
28	LHR	0% (DF)	3.6	3.4	3.2	3.6	3.4	3.2
29	LHR	0% (CJO)	3.0	2.8	2.6	3.0	2.8	2.6
	LHR	55%	2.7	2.5	2.3	2.9	2.7	2.5
31	CE	0% (DF)	4.2	4.4	4.6	4.2	4.4	4.6
32	CE	0% (CJO)	4.6	4.8	5.0	4.6	4.8	5.0
32	CE	35%	3.7	3.9	4.1	3.9	4.0	4.2

chamber and increase of same with conventional engine with pure diesel operation with advanced injection timing. Similar observations were made in Reference [21].

From the same table it is noticed that with vegetable oil operation, coolant load at full load operation decreased by 19% at recommended injection timing and 34% at optimized injection timing with engine with LHR combustion chamber, in comparison with conventional engine. This was due to improved combustion with provision of thermal insulation with engine with LHR combustion chamber. Similar trends were observed in Reference [22].

With maximum induction of methanol, coolant load at full load operation decreased by 11% at recommended injection timing and 14% at optimized injection timing, with engine with LHR combustion chamber in comparison with conventional engine. This was because of higher amount of methanol induction in engine with LHR combustion chamber leading to reduce coolant load, due to its high latent heat of evaporation. Similar trends were noticed in Reference [24].

From Table 7, it is noticed that coolant load was marginally higher with ethanol operation in comparison with methanol operation in both versions of the combustion chamber at recommended and optimum injection timings. This was due to high latent heat of evaporation of methanol.

From the same table, it is seen that coolant load increased marginally in conventional engine, while it decreased in engine with LHR combustion chamber with increasing of the injector opening pressure with test fuels. This was due to the fact that with an increase of injector opening pressure with conventional engine, increased nominal fuel spray velocity resulting in improved fuel–oxygen mixing with which gas temperatures increased. The reduction of coolant load in engine with LHR combustion chamber was not only due to the provision of the insulation but also it was due to improved fuel spray characteristics and increase of oxygen–fuel ratios causing decrease of gas temperatures and hence the coolant load.

Fig. 13 indicates that volumetric efficiency decreased in both versions of the combustion chamber with the dual fuel operation when compared with pure diesel operation on conventional engine, as percentage of alcohol induction increased, the amount of air admitted into the cylinder of the engine reduced.

Diesel fuel with conventional engine at optimized injection timing showed higher volumetric efficiency than other versions of the combustion chamber with test fuels. This is due to high cetane number and clean combustion at optimized injection timing with diesel fuel.

From Fig. 13, it is further understood that LHR version of the combustion chamber with methanol induction at recommended injection timing showed lower volumetric efficiency than other test fuels with other versions of the combustion chamber. This is due to (i) decrease of density of air with heat of the insulated components of LHR combustion chamber and (ii) replacement of air with methanol induction.

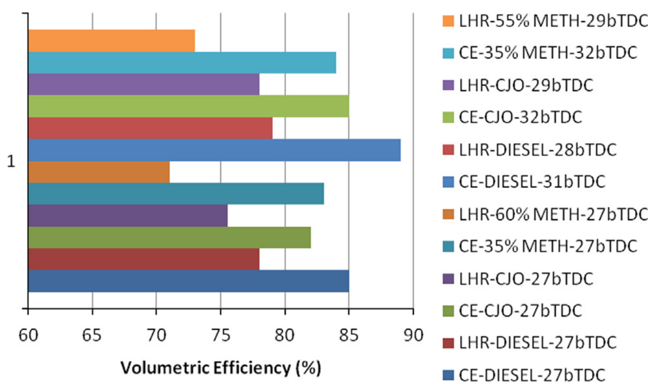

Fig. 13. Bar charts showing the variation of volumetric efficiency with test fuels in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

Table 8
Comparative data on volumetric efficiency (VE) at full load operation.

IT	Combustion chamber version	Alcohol induction on mass basis	Volumetric efficiency (%)					
			Methanol operation			Ethanol operation		
			Injector opening pressure (bar)			Injector opening pressure (bar)		
			190	230	270	190	230	270
27	CE	0% (DF)	85	86	87	85	86	87
		0% (CJO)	82	83	84	82	83	84
		35%	83	84	85	82	83	84
		40%	–	–	84	–	–	83
	LHR	0% (DF)	78	79	80	78	79	80
		0% (CJO)	75.5	76.5	77.5	75.5	76.5	77.5
		60%	71	72	73	70	71	72
28	LHR	0% (DF)	79	80	81	79	80	81
29	LHR	0% (CJO)	78	79	80	78	79	80
	LHR	55%	73	74	75	72	73	74
31	CE	0% (DF)	89	88	87	89	88	87
32	CE	0% (CJO)	85	84	83	85	84	83
32	CE	35%	84	85	86	83	84	85

Volumetric efficiency increased marginally with both versions of the combustion chamber with test fuels with advanced injection timing. This was due to decrease of combustion chamber wall temperatures with improved oxygen–fuel ratios.

From Table 8, it is observed that, with pure diesel operation, volumetric efficiency at full load operation decreased by 8% at recommended injection timing and 11% at optimized injection timing with engine with LHR combustion chamber, in comparison with conventional engine. This was because of increase of temperatures of insulated components in LHR engine, which heat the incoming charge to high temperatures and consequently the mass of air inducted in each cycle was lower. Similar observations were made in Reference [21].

From the same table, it is observed that, with vegetable oil operation, volumetric efficiency at full load operation decreased by 8% at recommended injection timing and 7% at optimized injection timing with engine with LHR combustion chamber, in comparison with conventional engine. Similar observations were noticed in Reference [22].

With maximum induction of methanol, volumetric efficiency at full load operation decreased by 14% at recommended injection timing and 11% at optimized injection timing, with engine with LHR combustion chamber in comparison with conventional engine. Similar observations were made in Reference [24].

From Table 8, it is evident that volumetric efficiency was higher with methanol operation in comparison with ethanol operation in both versions of the combustion chamber. This was due to decrease of gas temperatures because of high latent heat of evaporation of methanol.

Volumetric efficiency increased marginally with an increase of injector opening pressure in both versions of the combustion chamber with test fuels. This was due to improvement of air utilization and combustion with an increase of injector opening pressure. However, these variations were very small.

3.2.2. *Exhaust emissions*

Smoke and NO_x are exhaust emissions from diesel engine. They create health hazards, when they are inhaled, which cause severe headache, tuberculosis, lung cancer, nausea, respiratory problems, dizziness, skin cancer and hemorrhage [28,29]. The contaminated air containing carbon dioxide released from automobiles reach the ocean in the form of acid rain, thereby polluting water. Hence control of these emissions is an immediate task and important.

From Fig. 14, it is observed that engine with LHR combustion chamber with 55% methanol induction gave lower value of particulate emissions than other versions of the combustion

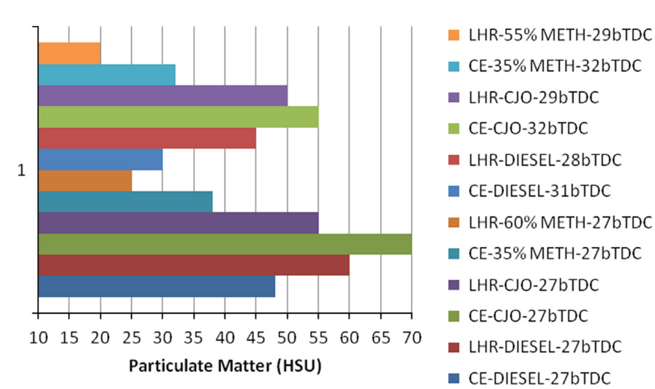

Fig. 14. Bar charts showing the variation of particulate matter in Hartridge Smoke Unit (HSU) with test fuels in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

chamber with other test fuels. The combustion of methanol is predominantly a process of hydroxylation and the chances of fuel cracking are negligible. Methanol does not contain carbon–carbon bonds and therefore cannot form any un-oxidized carbon particles or particulate emissions. One of the promising factors for reducing particulate emissions with the methanol is that it contains oxygen in its composition. Particulate emissions increase linearly with density of the fuel and increase of carbon to hydrogen atoms (C/H) ratio provided the equivalence ratio is not altered. This is because high values of C/H lead to more concentration of carbon dioxide, which would be further, reduced to carbon. Consequently, induction of alcohol reduced the quantity of carbon particles in the exhaust gases as the values of C/H for diesel fuel and methanol are 0.45 and 0.25 respectively. Crude vegetable oil operation on conventional engine gave higher particulate emissions. The combustion of injected fuel in case of vegetable oil operation is predominantly one of oxidation of products of destructive decomposition. In this case, there are greater chances of fuel cracking and forming carbon particles. C/H ratio for vegetable oil is 0.5, while density is 0.9 against 0.44 and 0.84 for diesel.

Particulate emissions decreased at optimum injection timing with test fuels. This was due to initiation of combustion at early period. This was due to increase of air entrainment, at the advanced injection timings, causing lower particulate emissions.

From Table 9, it is observed that, with pure diesel operation, particulate emissions at full load operation increased by 25% at recommended injection timing and 50% at optimized injection timing with engine with LHR combustion chamber, in comparison with conventional engine. This was due to fuel cracking at high temperatures. Similar trends were noticed in Reference [20].

From the same table, it is observed that, with vegetable oil operation, particulate emissions at full load operation decreased by 21% at recommended injection timing and 9% at optimized injection timing with engine with LHR combustion chamber, in comparison with conventional engine. This is due to improved combustion in the hot environment provided by LHR combustion chamber. Similar trends were observed in Reference [22].

Table 9
Comparative data on particulate matter in Hartridge Smoke Unit (HSU) at full load operation.

IT	Combustion chamber version	Alcohol induction on mass basis	Particulate matter (HSU)					
			Methanol operation			Ethanol operation		
			Injector opening pressure (bar)			Injector opening pressure (bar)		
			190	230	270	190	230	270
27	CE	0% (DF)	48	38	34	48	38	34
		0% (CJO)	70	65	60	70	65	60
		35%	38	33	28	42	37	32
		40%	–	–	25	–	–	30
	LHR	0% (DF)	60	55	50	60	55	50
		0% (CJO)	55	50	45	55	50	45
		60%	25	20	15	30	25	20
28	LHR	0% (DF)	45	40	35	45	40	35
29	LHR	0% (CJO)	50	45	40	50	45	40
	LHR	55%	20	17	13	25	22	18
31	CE	0% (DF)	30	30	35	30	30	35
32	CE	0% (CJO)	55	60	65	55	60	65
	CE	35%	32	28	24	37	33	29

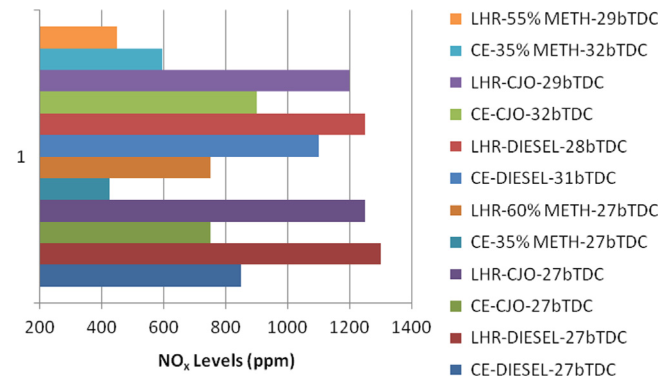

Fig. 15. Bar charts showing the variation of nitrogen oxide (NO_x) levels with test fuels in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

With maximum induction of methanol, engine with LHR combustion chamber decreased particulate emissions by 34% at recommended injection timing and 50% at optimized injection timing in comparison with conventional engine. This is due to higher amount of methanol induction in engine with LHR combustion chamber which caused improved combustion with reduction of fuel cracking temperatures.

Particulate matter emissions were marginally lower with methanol operation in comparison with ethanol operation in both versions of the engine as C/H ratio of methanol (0.25) is lower than ethanol (0.33).

Particulate emissions further decreased with an increase of injector opening pressure in both versions of the combustion chamber, as it is noticed from Table 9, due to efficient combustion at higher injector opening pressures, which improved the atomization with the reduction of mean diameter of the fuel particle.

From Fig. 15, it is noticed that 35% methanol induction in conventional engine gave lower value of NO_x emissions compared with other versions of the combustion chamber and test fuels. This was due to reduction of gas temperatures with methanol induction in conventional engine.

From the same figure, it is noticed that, diesel operation on engine with LHR combustion chamber at recommended injection timing gave higher value of NO_x emissions than other versions of the combustion chamber with test fuels. This was due to increase of gas temperatures with reduction of ignition delay. This was also because of high calorific value of diesel fuel.

NO_x levels further decreased with the increase of methanol induction in both versions of the combustion chamber. This was due to reduction of gas temperatures as methanol has high latent heat of evaporation.

From the same figure, it was further observed that with increasing the injection advance resulted in higher combustion temperatures and increase of resident time leading to produce more NO_x concentration in the exhaust of conventional engine with test fuels.

At the optimum injection timing, engine with LHR combustion chamber, with test fuels recorded lower NO_x emissions, at full load operation compared to the same version of the engine at the recommended injection timing. This was due to decrease of combustion temperatures with improved oxygen–fuel ratios.

From Table 10, it is observed that, with pure diesel operation, LHR version of the combustion chamber gave higher value of NO_x emissions by 53% at recommended injection timing and 14% at

Table 10
Comparative data on NO_x levels at full load operation.

IT	Combustion chamber version	Alcohol induction on mass basis	NO_x levels (ppm)					
			Methanol operation			Ethanol operation		
			Injector opening pressure (bar)			Injector opening pressure (bar)		
			190	230	270	190	230	270
27	CE	0% (DF)	850	900	950	850	900	950
		0% (CJO)	750	800	850	750	800	850
		35%	425	475	525	475	525	575
		40%	–	–	475	–	–	525
	LHR	0% (DF)	1300	1250	1200	1300	1250	1200
		0% (CJO)	1250	1200	1150	1250	1200	1150
		60%	750	800	850	800	850	900
28	LHR	0% (DF)	1250	1200	1150	1250	1200	1150
29	LHR	0% (CJO)	1200	1150	1100	1200	1150	1100
	LHR	55%	450	400	350	525	475	425
31	CE	0% (DF)	1100	1150	1200	1100	1150	1200
32	CE	0% (CJO)	900	950	1000	900	950	1000
32	CE	35%	595	650	700	645	700	750

optimized injection timing in comparison with conventional engine. This was due to increase of gas temperatures with reduction of ignition delay. Similar trends were noticed in Reference [20].

From the same table, it is noticed that, with vegetable oil operation, LHR version of the combustion chamber gave higher value of NO_x emissions by 66% at recommended injection timing and 33% at optimized injection timing in comparison with conventional engine. This was due to increase of gas temperatures. Similar observations were noticed in Reference [22].

With maximum induction of methanol, engine with LHR combustion chamber gave higher value of NO_x emissions by 76% at recommended injection timing and decreased the same by 24% in comparison with conventional engine. This was due to increase of gas temperature at recommend injection and decrease the same at optimized injection timing with engine with LHR combustion chamber. Similar trends were noticed in Reference [24].

NO_x levels were lower with methanol operation when compared with ethanol operation on both versions of the combustion chamber. This was due to decrease of gas temperatures because of high latent heat of evaporation of methanol.

However, they decreased with an increase of injector opening pressure in engine with LHR combustion chamber while they increased in conventional engine. This was due to decrease of gas temperatures with engine with LHR combustion chamber and increase of the same with increase of injector opening pressure.

These aldehydes are responsible for pungent smell of the engine and affect the human beings when inhaled in large quantities. The volatile aldehydes are eye and respiratory tract irritants. Though Government legislation has not been pronounced regarding the control of aldehyde emissions, when more and more alcohol engines are coming to existence, severe measures to control the aldehydes emitted out through the exhaust of the alcohol run engines will have to be taken as a serious view.

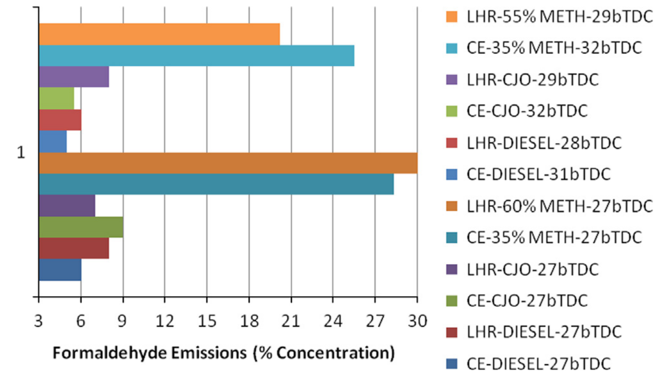

Fig. 16. Bar charts showing the variation of formaldehyde emissions with test fuels in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

Table 11
Comparative data on formaldehyde (% concentration) emissions at full load operation.

IT	Combustion chamber version	Alcohol induction on mass basis	Formaldehyde emissions (% concentration)					
			Methanol operation			Ethanol operation		
			Injector opening pressure (bar)			Injector opening pressure (bar)		
			190	230	270	190	230	270
27	CE	0% (DF)	6.0	5.5	5.0	6.0	5.5	5.0
		0% (CJO)	9.0	8.1	6.9	9.0	8.1	6.9
		35%	28.3	26.2	24.1	18.3	16.3	14.2
		40%	–	–	26.4	–	–	16.4
	LHR	0% (DF)	8.0	6.6	6.0	7.0	6.8	6.6
		0% (CJO)	7.0	6.8	6.6	7.0	6.8	6.6
		60%	30.2	28.2	26.6	24.3	22.1	20.4
28	LHR	0% (DF)	6	5	4	6	5	4
29	LHR	0% (CJO)	5.5	4.5	3.5	5.5	4.5	3.5
	LHR	55%	20.2	18.2	16.4	15.5	13.6	11.5
31	CE	0% (DF)	5	4	3	5	4	3
32	CE	0% (CJO)	6.2	5.8	5.4	6.2	5.8	5.4
32	CE	35%	25.5	23.3	21.5	13.0	11.4	9.5

It could be seen from Fig. 16, that formaldehyde emissions were higher with methanol induction in both versions of the combustion chamber. This was due to oxidation reaction of methanol with hydrocarbon fuels. This was due to partial oxidation compared to pure diesel. The low combustion temperature leads to produce partially oxidized carbonyl (aldehyde) compounds with methanol blended gasoline.

Formaldehyde emissions were quiet low with non-alcoholic fuels with engine with LHR combustion chamber at their optimized injection timings as noticed from Fig. 16.

Formaldehyde emissions decreased marginally with advanced injection timing with test fuels. This was because of improved combustion with reduction intermediate compounds.

From Table 11, it is observed that, with pure diesel operation, LHR version of the combustion chamber gave higher value of formaldehyde emissions by 33% at recommended injection timing and 25% at optimized injection timing in comparison with conventional engine. This was due to reduction of ignition delay with formation of intermediate compounds like formaldehyde during combustion.

From the same table, it is further noticed that, with vegetable oil operation, LHR version of the combustion chamber gave lower value of formaldehyde emissions by 18% at recommended injection timing and 11% at optimized injection timing in comparison with conventional engine. This was due to improved combustion with reduction of intermediate compounds.

With maximum induction of methanol, engine with LHR combustion chamber gave higher value of formaldehyde emissions by 7% at recommended injection timing and decreased the same by 25% in comparison with conventional engine. This was due to higher amount of substitution (60%) of methanol in engine with LHR combustion chamber at recommended injection timing and improved combustion at optimized injection timing. Similar trends were noticed in Reference [24].

Hot environment of LHR engine completed combustion reactions and reduced the emissions of intermediate compounds, formaldehydes. Hence it is concluded that engine with LHR combustion chamber was more suitable for carbureted alcohol in comparison with pure diesel operation.

Formaldehyde emissions were higher with methanol operation when compared with ethanol operation on both versions of the combustion chamber. This was due to oxidation reaction with methanol forming formaldehydes.

Increase of injector opening pressure also improved the combustion performance in both versions of the combustion chamber leading to reduction in the intermediate compounds like formaldehydes with test fuels.

It is observed from Fig. 17, that acetaldehyde emissions were higher with methanol induction in both versions of the combustion chamber. This was due to oxidation reaction of methanol with hydro-carbon fuels.

Acetaldehyde emissions were quiet low with non-alcoholic fuels with engine with LHR combustion chamber at their optimized injection timings as noticed from Fig. 17.

Acetaldehyde emissions decreased marginally with advanced injection timing with test fuels. This was due to initiation of combustion at early period with improved oxygen-entrainment.

From Table 12, it is observed that, with pure diesel operation, LHR version of the combustion chamber gave higher value of acetaldehyde emissions by 38% at recommended injection timing

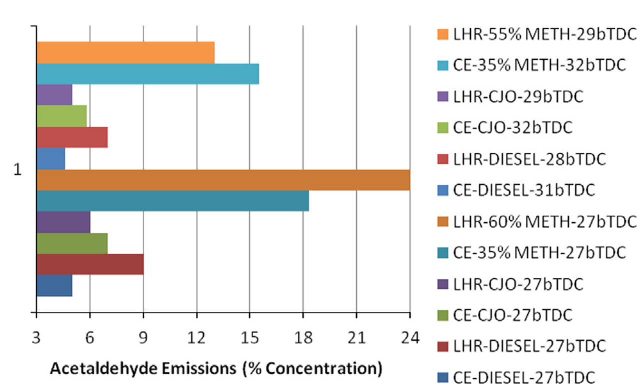

Fig. 17. Bar charts showing the variation of acetaldehyde emissions with test fuels in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

Table 12
Comparative data on acetaldehyde emissions (% concentration) at full load operation.

IT	Combustion chamber version	Alcohol induction on mass basis	Acetaldehyde emissions (% concentration)					
			Methanol operation			Ethanol operation		
			Injector opening pressure (bar)			Injector opening pressure (bar)		
			190	230	270	190	230	270
27	CE	0% (DF)	5	4.5	4	7	6	4.9
		0% (CJO)	7	6	4.9	8	7	5.9
		35%	18.3	16.4	14.7	28.3	26.5	24.5
		40%	–	–	16.5	–	–	26.7
	LHR	0% (DF)	9	8	7	9	8	7
		0% (CJO)	6	5	4	6	5	4
		60%	24.3	22.7	20.5	30.2	28.6	26.6
28	LHR	0% (DF)	7	6	5	7	6	5
29	LHR	0% (CJO)	5	4	3	5	4	3
	LHR	0% (55%)	13	11.4	9.4	20.2	18.3	16.4
31	CE	0% (DF)	4.6	4.4	4.2	4.6	4.4	4.2
32	CE	0% (CJO)	5.8	5.6	5.4	7.8	6.8	6.4
32	CE	35%	15.5	13.7	11.5	25.5	23.5	21.5

and 40% at optimized injection timing in comparison with conventional engine. This was due to reduction of ignition delay with formation of intermediate compounds like acetaldehydes during combustion.

From the same table, it is noticed that, with vegetable oil operation, LHR version of the combustion chamber gave lower value of acetaldehyde emissions by 12% at recommended injection timing and 14% at optimized injection timing in comparison with CE. This was due to improved combustion with reduction of intermediate compounds.

With maximum induction of methanol, engine with LHR combustion chamber gave higher value of acetaldehyde emissions by 33% at recommended injection timing and decreased the same by 13% in comparison with conventional engine. This was due to higher amount of substitution (60%) of methanol in engine with LHR combustion chamber at recommended injection timing and improved combustion at optimized injection timing. Similar trends were noticed in Reference [24].

Acetaldehyde emissions were higher with ethanol operation when compared with methanol operation on both versions of the combustion chamber. This was due to oxidation reaction with ethanol forming acetaldehydes.

Increase of injector opening pressure also improved the combustion performance in both versions of the combustion chamber leading to reduction in the intermediate compounds like acetaldehydes with test fuels.

3.2.3. Combustion characteristics

From Fig. 18, it is noticed that with 55% methanol induction in engine with LHR combustion chamber at its optimized injection

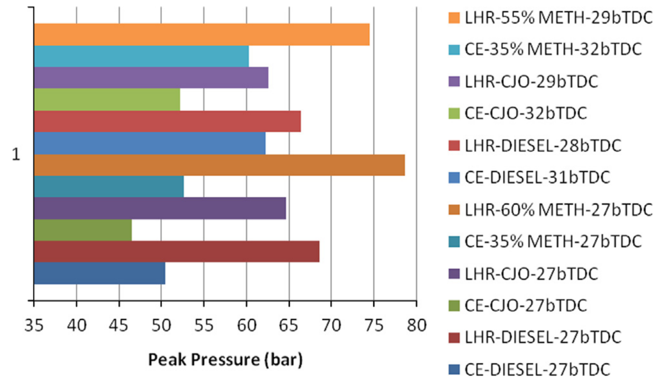

Fig. 18. Bar charts showing the variation of peak pressure with test fuels in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

Table 13
Comparative data on peak pressure at full load operation.

IT	Combustion chamber version	Alcohol induction on mass basis	Peak pressure (bar)					
			Methanol operation			Ethanol operation		
			Injector opening pressure (bar)			Injector opening pressure (bar)		
			190	230	270	190	230	270
27	CE	0% (DF)	50.4	51.7	53.5	50.4	51.7	53.5
		0% (CJO)	46.5	47.5	48.5	46.5	47.5	48.5
		35%	52.6	53.7	54.8	53.6	54.7	55.8
		40%	–	–	56.4	–	–	57.6
	LHR	0% (DF)	68.6	66.6	64.6	68.6	66.6	64.6
		0% (CJO)	64.6	62.6	60.6	64.6	62.6	60.6
		60%	78.6	76.2	74.2	76.6	74.6	72.5
28	LHR	0% (DF)	66.4	65.4	64.5	66.4	65.4	64.5
29	LHR	0% (CJO)	62.5	60.5	58.7	62.5	60.5	58.7
	LHR	55%	74.4	72.2	70.2	72.2	70.2	68.2
31	CE	0% (DF)	62.2	62.6	63.2	62.4	60.6	58.6
32	CE	0% (CJO)	52.2	54.2	56.2	54.5	53.2	52.1
	CE	35%	60.2	61.2	62.2	63.1	65.1	66.2

timing gave higher value of peak pressure in comparison with other versions of the combustion chamber with test fuels. This was due to improved combustion with improved homogeneity of the mixture approaching constant volume combustion.

Crude jatropha oil with conventional engine at recommended injection timing gave lower value of peak pressure when compared with other versions of the combustion chamber with test fuels. This was due to low calorific value of the fuel with high duration of combustion with retarded heat release rates.

The value of peak pressure increased with advancing of the injection timing with conventional engine and decreased marginally with engine with LHR combustion chamber with test fuels. This was due to accumulated and sudden explosion of the fuel with advanced injection timing with conventional engine. Improved combustion with improved oxygen–fuel ratios resulted in lower peak pressure with engine with LHR combustion chamber. This was also due to reduction of alcohol induction at optimum injection timing in comparison with recommended injection timing.

From Table 13, it is observed that, with pure diesel operation, LHR version of the combustion chamber gave higher value of peak pressure by 36% at recommended injection timing and 6% at optimized injection timing in comparison with conventional engine. This was due to reduction of ignition delay with increased mass flow rate of burning.

From the same table, it is further noticed that, with vegetable oil operation, LHR version of the combustion chamber gave higher value of peak pressure by 39% at recommended injection timing and 20% at optimized injection timing in comparison with conventional engine. This was due to improved combustion of vegetable oil with engine with LHR combustion. With conventional engine, this was due to increase of ignition delay, as vegetable oil requires large duration of combustion, meanwhile the piston started making downward motion thus increasing volume when the combustion takes place in conventional engine. Similar trends were noticed in Reference [22].

With maximum induction of methanol, engine with LHR combustion chamber gave higher value of peak pressure by 50% at recommended injection timing and 23% at optimized injection timing in comparison with conventional engine. This was due to higher amount of substitution (60%) of methanol in engine with LHR combustion chamber at recommended injection timing and improved combustion at optimized injection timing. Similar trends were observed in Reference [24].

Conventional engine with ethanol induction gave marginally higher value of peak pressure in comparison with methanol induction in the same version of the engine at recommended and optimized injection timing. This was due to higher calorific value of the ethanol in comparison with methanol.

The engine with LHR combustion chamber with methanol induction gave marginally higher value of peak pressure in comparison with ethanol induction in the same version of the engine at recommended and optimized injection timing. This was due to improved evaporation rate of methanol in hot environment provided by LHR combustion chamber.

Peak pressures increased with conventional engine with an increase of injector opening pressure with the test fuels. This may be due to smaller sauter mean diameter, shorter breakup length, improved dispersion and improved spray and atomization characteristics with conventional engine. This improves combustion rate in the premixed combustion phase. Peak pressure decreased with engine with LHR combustion chamber with test fuels with

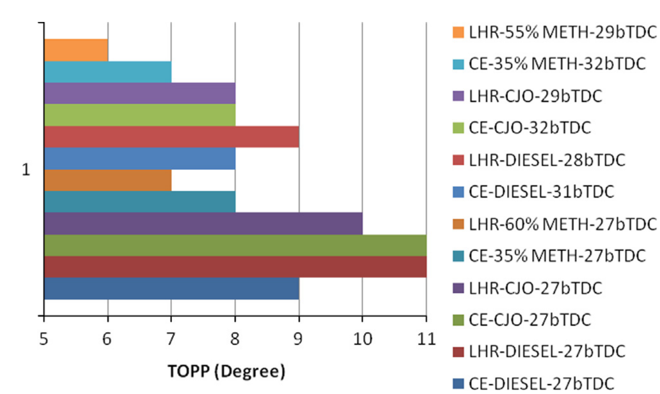

Fig. 19. Bar charts showing the variation of time of occurrence of peak pressure (TOPP) with test fuels in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

increase of injector opening pressure due to decrease of gas temperature with improved combustion.

From Fig. 19, it is observed that engine with LHR combustion chamber with 55% methanol induction at its optimized injection timing gave lower value (closer to TDC) of time of occurrence of peak pressure (TOPP) in comparison with other versions of the combustion chamber with other test fuels. This was once again confirmed by the observation of higher peak pressure and lower TOPP in engine with LHR combustion chamber that the performance of engine with LHR combustion chamber with 55% methanol induction at its optimized injection timing improved over other versions of the combustion chamber with other test fuels.

TOPP decreased with advancement of the injection timing with both versions of the combustion chamber with test fuels. This was because of high duration of combustion of low cetane fuels leading to burn near TDC.

Vegetable oil operation with conventional engine at recommended injection timing gave higher TOPP in comparison with other versions of the combustion chamber with test fuels. This was due to increase of combustion duration and retarded heat release rate. This showed that performance deteriorated with vegetable oil operation on conventional engine.

Diesel operation with engine with LHR combustion chamber at recommended injection timing also gave higher TOPP in comparison with other versions of the combustion chamber. This was due to continued combustion of diesel fuel even after piston started making downward stroke.

From Table 14, it is observed that, with pure diesel operation, LHR version of the combustion chamber increased TOPP by 22% at recommended injection timing and 12% at optimized injection timing in comparison with conventional engine. This was due to continued combustion of the diesel fuel even after piston started executing power stroke. Similar observations were made in Reference [20].

From the same table, it is further noticed that, with vegetable oil operation, LHR version of the combustion chamber gave lower TOPP by 9% at recommended injection timing and comparable at optimized injection timing in comparison with conventional engine. This was due to high duration of combustion and high viscosity with crude vegetable oil in the environment of conventional engine, while improved combustion of vegetable oil with engine with LHR combustion chamber. Similar observations were noticed in Reference [22].

With maximum induction of methanol, engine with LHR combustion chamber decreased TOPP by 12% at recommended injection timing and 14% at optimized injection timing in comparison with conventional engine. This was due to higher amount of substitution (60%) of methanol in engine with LHR combustion chamber at recommended injection timing and improved combustion at optimized injection timing. Similar trends were observed in Reference [24].

TOPP with methanol operation was comparable with ethanol operation on both versions of the combustion chamber at recommended and optimized injection timing. This was due to higher calorific value of the ethanol in comparison with methanol.

TOPP decreased marginally with both versions of the combustion chamber with test fuels with an increase of injection pressure at recommended and optimized injection timings. This was due to improved spray characteristics of the fuel.

Maximum rate of pressure rise (MRPR) followed the similar trends of peak pressure in both versions of the combustion chamber with test fuels.

From Fig. 20, engine with LHR combustion chamber with 55% methanol induction at its optimum injection timing gave higher value of MRPR in comparison with other versions of the combustion chamber with other test fuels. This was due to improved evaporation rate of methanol with hot insulated components of LHR combustion chamber.

Conventional engine with crude vegetable oil operation at recommended injection timing gave lower MRPR in comparison with other configurations of combustion chamber with other test fuels. This was due to retarded heat release rate, higher duration of combustion of vegetable oil in conventional engine in addition to its lower calorific value.

MRPR increased marginally with advanced injection timing with test fuels. This was because of initiation and continued combustion at early period.

Table 14
Comparative data on time of occurrence of peak pressure (TOPP) at full load operation.

IT	Combustion chamber version	Alcohol induction on mass basis	TOPP (degree)					
			Methanol operation			Ethanol operation		
			Injector opening pressure (bar)			Injector opening pressure (bar)		
			190	230	270	190	230	270
27	CE	0% (DF)	9	9	8	9	9	8
		0% (CJO)	11	11	11	11	11	11
		35%	8	8	7	8	7	6
		40%	–	–	7	–	–	6
	LHR	0% (DF)	11	10	10	11	10	10
		0% (CJO)	10	10	9	10	10	9
		60%	7	7	6	7	7	6
28	LHR	0% (DF)	9	9	9	9	9	9
29	LHR	0% (CJO)	8	8	8	8	8	8
	LHR	55%	6	6	6	6	6	6
31	CE	0% (DF)	8	9	9	8	9	9
32	CE	0% (CJO)	8	9	9	8	9	9
32	CE	35%	8	7	7	7	7	7

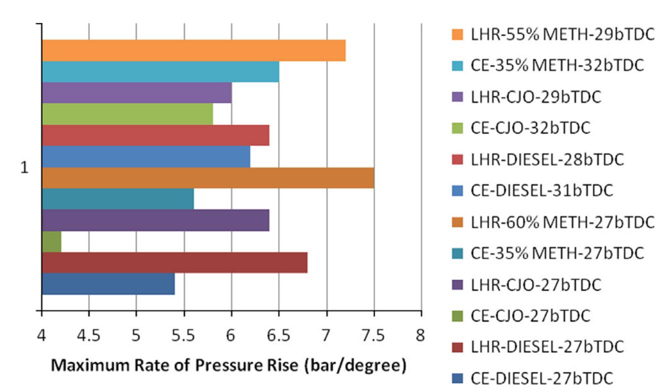

Fig. 20. Bar charts showing the variation of maximum rate of pressure rise (MRPR) with test fuels in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

Table 15
Comparative data on maximum rate of pressure rise (MRPR) at full load operation.

IT	Combustion chamber version	Alcohol induction on mass basis	MRPR (bar/degree)					
			Methanol operation			Ethanol operation		
			Injector opening pressure (bar)			Injector opening pressure (bar)		
			190	230	270	190	230	270
27	CE	0% (DF)	5.4	5.6	6.0	5.4	5.6	6.0
		0% (CJO)	4.2	4.4	4.6	4.2	4.4	4.6
		35%	5.6	5.8	6.0	5.8	6.0	6.2
		40%			6.2	–	–	6.4
	LHR	0% (DF)	6.8	6.6	6.4	6.8	6.6	6.4
		0% (CJO)	6.4	6.2	6.0	6.4	6.2	6.0
		60%	7.5	7.3	7.1	7.2	7.0	6.8
28	LHR	0% (DF)	6.4	6.0	5.6	6.4	6.0	5.6
29	LHR	0% (CJO)	6.0	5.8	5.5	6.0	5.8	5.5
	LHR	55%	7.2	7.0	6.8	7.0	6.8	6.6
31	CE	0% (DF)	6.2	6.0	5.8	6.2	6.0	5.8
32	CE	0% (CJO)	5.8	6.0	6.2	5.8	6.0	6.2
	CE	35%	6.5	6.7	6.9	6.5	6.7	6.9

From Table 15, it is observed that, with pure diesel operation, LHR version of the combustion chamber increased MRPR by 27% at recommended injection timing and comparable at optimized injection timing in comparison with CE. This was due to reduction of ignition delay with diesel. Similar observations were made in Reference [20].

From Table 15, it is further noticed that, with vegetable oil operation, LHR version of the combustion chamber increased MRPR by 52% at recommended injection timing and 23% at optimized injection timing in comparison with conventional engine. This was due to improved combustion of vegetable oil with engine in the hot environment provided by the engine with LHR combustion chamber. Similar observations were noticed in Reference [22].

With maximum induction of methanol, engine with LHR combustion chamber increased MRPR by 50% at recommended injection timing and 10% at optimized injection timing in comparison with conventional engine. This was due to higher amount of substitution (60%) of methanol in engine with LHR combustion chamber at recommended injection timing and improved combustion at optimized injection timing. Similar observations were noticed in Reference [24].

For the same amount of alcohol induction, higher MRPR was observed with conventional engine with ethanol induction, while engine with LHR combustion chamber with methanol induction recorded higher MRPR at recommended injection timing and optimized injection timing. This was due to higher calorific value of the ethanol in comparison with methanol and good evaporation characteristics of methanol when compared with ethanol.

MRPR increased marginally with conventional engine and decreased the same with engine with LHR combustion chamber with test fuels with an increase of injector opening pressure at recommended and optimized injection timings. This was due to improved spray characteristics of the fuel. MRPR was greater for dual fuel operation on engine with LHR combustion chamber due to evaporation of higher amount of alcohol substitution.

4. Summary

1. With vegetable oil operation, the performance of the engine with LHR combustion chamber improved performance in comparison with conventional engine at recommended and optimized injection timings.
2. The maximum induction of alcohol (ethanol/methanol) was found to be 35% with CE, while it was 60% with engine with LHR combustion chamber at recommended injection timing on mass basis of vegetable oil at full load operation.
3. The maximum induction of alcohol (ethanol/methanol) was observed to be 35% with conventional engine, while it was 55% with engine with LHR combustion chamber at optimized injection timing on mass basis of vegetable oil at full load operation.
4. With maximum induction of alcohol, at recommended injection timing and optimized injection timings, the performance of the engine with LHR combustion chamber improved in comparison with conventional engine.
5. Methanol induction improved performance with engine with LHR combustion chamber, while ethanol induction showed improved performance with conventional engine at recommended injection timing and optimized injection timing.
6. Methanol operation increased formaldehyde emissions while ethanol operation increased acetaldehyde emissions drastically on both versions of the combustion chamber at recommended and optimized injection timings in comparison with diesel operation on conventional engine.
7. At an injector opening pressure of 190 bar, with maximum induction of methanol, engine with LHR combustion chamber at its optimum injection timing, increased peak brake thermal efficiency by 13%, at full load operation-decreased brake specific energy consumption by 4%, decreased exhaust gas temperature by 31%, decreased coolant load by 10%, decreased volumetric efficiency by 6%, decreased particulate emissions by 60%, decreased NO_x levels by 62%; increased formaldehyde and acetaldehyde emissions drastically, increased peak pressure by 19% and increased maximum rate of pressure rise by 20% in comparison with engine with LHR combustion chamber with pure vegetable oil operation at its optimum injection timing.

4.1. Research findings

Investigations were carried out to evaluate the performance with conventional engine and engine with LHR combustion chamber with test fuels of pure diesel, vegetable oil and injected vegetable oil along with carbureted methanol and carbureted ethanol. Comparative studies were made with engine with LHR combustion chamber with test fuels with data of conventional engine with test fuels. Comparative studies were also made with methanol operation with data of ethanol operation on both versions of the combustion chamber.

4.2. Future scope of the work

1. Investigations can also be carried out to evaluate the performance of indirect injection diesel engine with test fuels.
2. Investigations can also be carried out to evaluate the performance of direct-injection diesel engine with carbureted alcohol with wide variety of vegetable oil/biodiesel.
3. A suitable catalytic converter can be designed to control formaldehyde and acetaldehyde emissions from engine run with alcohol with less expensive and easily available catalysts. Sponge iron and manganese ore can be used as catalysts. Air injection into the catalytic chamber can further reduce

formaldehyde levels and acetaldehyde levels [30]. Diameter of the pipe through which exhaust gas enters into the catalytic chamber is to be calculated by knowing the discharge of the fuel gases from the engine and velocity of the exhaust gases (3–4 m/s). The length of the catalytic chamber is to be calculated knowing the pressure drop [31].

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